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RESEARCH MEMORANDUM

OVER-ALL PERFORMANCE OF J35-A-23 TWO-STAGE TURBINE

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OVER-ALL PERFORMANCE OF J35-A-23 TWO-STAGE TURBINE

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SUMMARY

An investigation was conducted to determine the over-all performance of a two-stage turbine from a J35-A-23 turbojet engine.

The results of this investigation showed that limiting blade loading occurred in the second-stage rotor, restricting the equivalent work to approximately 95 percent of the design value.

The turbine brake internal efficiency at the design operating conditions was approximately 0.75. A maximum brake internal efficiency between 0.81 and 0.82 occurred at a pressure ratio of approximately 2.6 and equivalent design speed. The equivalent weight flow at equivalent design speed and pressure ratio was approximately 106 percent of the design value.

INTRODUCTION

As part of a general study of high-work-output, low-speed multistage turbines, an investigation was undertaken at the NACA Lewis laboratory to determine the over-all performance characteristics of a two-stage turbine from the J35-A-23 turbojet engine.

The turbine was operated at a constant inlet stagnation pressure of 40.5 inches of mercury absolute and an inlet stagnation temperature of 700° R over a range of pressure ratio for equivalent rotative speeds of 20, 40, 60, 70, 80, 90, 100, 110, 120, and 130 percent of the design value.

The over-all performance is presented in terms of brake internal efficiency and equivalent work (each based upon torque measurements), equivalent pressure ratio, equivalent rotor speed, and equivalent weight flow.

An approximate method (based on an average equilibrium value of specific heat) of determining the equivalent design parameters of the turbine is presented and briefly discussed.

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SYMBOLS

A	area, (sq ft)
E	enthalpy drop based on torque measurements, (Btu/lb)
g	acceleration due to gravity, (32.174 ft/sec^2)
J	mechanical equivalent of heat, (778 ft-lb/Btu)
N	rotational speed, (rpm)
p	static pressure, (lb/sq ft)
p'	stagnation pressure, (lb/sq ft)
p'_x	static pressure plus Δp corresponding to axial component of velocity, (lb/sq ft)
R	gas constant for air, (53.3 ft-lb/(lb)($^{\circ}$R))
T'	stagnation temperature, ($^{\circ}$R)
U	blade velocity, (ft/sec)
V	absolute gas velocity, (ft/sec)
w	weight flow, (lb/sec)
β	function of γ, $\dfrac{\gamma_0}{\gamma_e}\left[\dfrac{\left(\dfrac{\gamma_e+1}{2}\right)^{\frac{\gamma_e}{\gamma_e-1}}}{\left(\dfrac{\gamma_0+1}{2}\right)^{\frac{\gamma_0}{\gamma_0-1}}}\right]$
γ	ratio of specific heats, c_p/c_v
δ	ratio of inlet-air pressure to NACA standard sea-level pressure, (p'_1/29.92 in. Hg abs.)
η_i	brake internal efficiency defined as the ratio of actual turbine work based on torque measurements to ideal turbine work based on inlet stagnation pressure, and outlet static pressure plus Δp corresponding to the axial component of velocity.
θ_{cr}	squared ratio of critical velocity to critical velocity at NACA standard sea-level temperature (518.4° R), $\left(\dfrac{V_{cr,e}}{V_{cr,0}}\right)^2$

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ρ gas density, (lb/cu ft)

τ torque, (ft-lb)

$\frac{wN\beta}{60\delta}$ parameter based on equivalent weight flow and equivalent rotational speed

Subscripts:

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1 turbine inlet measuring station

2 turbine outlet measuring station

0 NACA standard sea-level conditions

cr critical

e engine operating conditions

u tangential

x axial

APPARATUS

Turbine. - The two-stage turbine for a J35-A-23 turbojet engine was designed for the following conditions:

	Actual (engine operating conditions)	Equivalent (NACA standard sea-level conditions)
Work, Btu/lb	131.2	32.4
Weight flow, lb/sec	150.0	37.58
Rotative speed, rpm	6100	3035
Inlet temperature, ^{o}R	2160	518.4

The design of the first and second stages resulted in 44.4 and 46.0 percent reaction, respectively, at the mean radius. The tip diameter of the turbine is constant at 33.5 inches; the annular area increases through the turbine (the inner shroud has a cone half-angle of 11^{o}). The mean hub-tip radius ratio for the first- and second-stage rotors is 0.795 and 0.746, respectively.

Dynamometers. - Two cradled dynamometers of the eddy-current wet-gap type connected in tandem were used to absorb the output of the turbine. The turbine torque output was measured by means of an NACA balanced-diaphram type thrust meter.

Test installation. - The experimental setup of the turbine is shown in figure 1. Air flow to the unit was supplied by the laboratory combustion-air system at approximately 110 inches of mercury absolute and passed through a submerged orifice. After metering, the air was throttled to approximately 40.5 inches of mercury absolute and heated by means of two standard jet-engine burners to approximately 700° R. The air flow was divided and entered a plenum chamber (fig. 2) (which replaced the normal combustion can assembly of the engine) through two openings 180° apart and at right angles to the turbine shaft. The air then passed through a screen and into 10 standard transition sections, each of which supplied air to a segment of the first-stage stator. The air passed through the first and second stages of the turbine into the tail cone where it was discharged into the laboratory exhaust facilities.

INSTRUMENTATION

Turbine weight flow. - The air weight flow through the turbine was measured by a submerged ASME flange-tap flat-plate orifice. Fuel flow to the burners was measured by rotameters in the fuel line.

Turbine instrumentation. - The gas state in the turbine was measured at the two axial stations shown in figure 2. The turbine inlet conditions were measured by means of a combination probe consisting of a shielded total-pressure tube and a calibrated thermocouple, and two static-pressure taps in each of the 10 standard transition sections. The turbine exit conditions were determined by means of four thermocouple rakes, each consisting of five thermocouples located at the center of five equal annular areas; five shielded total-pressure probes, located at different circumferential positions and radii corresponding to the center of five equal annular areas; and four static-pressure taps on both the inner and the outer shroud.

Precision. - The precision of the measurements is estimated to be within the following limits:

Temperature, °R	±1.0
Pressure, in. Hg	±0.05
Air weight flow, percent	±1.0
Rotor speed, percent	±0.5
Torque, percent	±0.5

The cumulative effect on calculated turbine efficiency, using measurements of the foregoing precision, would give a maximum error of ±2.0 percent.

METHODS AND PROCEDURE

In order to determine the equivalent design conditions for this turbine, the variation in the ratio of specific heats from the design turbine inlet temperature to standard atmospheric temperature should be considered. An approximate method of determining the equivalent design conditions was used in this particular investigation. This method is based on the critical velocity determined from the turbine inlet stagnation temperature and an average equilibrium value of γ of the flow
2200 through the turbine. Derivation of this method is presented in the appendix.

The turbine was operated with an inlet pressure of approximately 40.5 inches of mercury absolute and an inlet temperature of 700° R for equivalent rotative speeds of 20, 40, 60, 70, 80, 90, 100, 110, 120, and 130 percent of the design value for pressure ratios $p'_1/p'_{x,2}$ of 1.4 to 4.08. The work output and brake internal efficiency are based on measured torque values.

RESULTS AND DISCUSSION

Because the specific purpose of this investigation was to determine the over-all performance of the J35-A-23 two-stage turbine, the performance data are presented in a form that readily shows the relations among the following turbine parameters: equivalent weight flow, equivalent total-pressure ratio, equivalent work output, brake internal efficiency, equivalent rotative speed, and equivalent torque output of the unit.

The over-all performance of the turbine is presented in figure 3 in terms of equivalent work and a weight flow parameter $wN\beta/60\delta$ for lines of constant equivalent speed, pressure ratio $p'_1/p'_{x,2}$, and brake internal efficiency. The pressure ratio $p'_1/p'_{x,2}$ is defined as the ratio of the turbine inlet stagnation pressure p'_1 to the static pressure plus the Δp corresponding to the axial component of the velocity at the turbine exit $p'_{x,2}$. The pressure $p'_{x,2}$ is calculated from a measured stagnation pressure, static pressure, stagnation temperature, and weight flow.

A maximum brake internal efficiency between 0.81 and 0.82 was reached at a pressure ratio of approximately 2.6 and equivalent design speed. At the equivalent design pressure ratio of 4.03 and the equivalent design speed, the brake internal efficiency was approximately 0.75. The design pressure ratio $(p'_1/p'_{x,2})_e$ was determined from the design velocity diagram for the mean radius at the exit of the second-stage rotor by adding the Δp corresponding to the axial component of velocity to the static pressure. The equivalent design pressure ratio $(p'_1/p'_{x,2})_0$ was then computed considering the change in γ.

From the equivalent design values of E/θ_{cr} and $wN\beta/60\delta$ indicated in figure 3, it may be seen that design work was not obtained at design operating conditions, the maximum work obtained being approximately 95 percent of design equivalent work.

The variation of equivalent weight flow with over-all pressure ratio over a range of equivalent rotor speeds is shown in figure 4. The figure indicates that for equivalent rotative speeds of 60 percent of design or higher the operating pressure ratio is high enough to choke the turbine. The value of choking weight flow, however, is different for each rotor speed, which indicates that the turbine chokes somewhere downstream of the first-stage stator. The actual value of the equivalent weight flow obtained at the equivalent design speed and design pressure ratio is approximately 106 percent of the design value. 2208

Figure 5 shows the variation of equivalent torque output as a function of total to static pressure ratio p'_1/p_2 for various equivalent rotor speeds. At the equivalent design total to static pressure ratio and equivalent rotor speeds of 100 to 130 percent, the slope of the torque curves approaches zero, which indicates that further increase in pressure ratio would yield little, if any, increase in torque output at any particular speed from 100 to 130 percent of design. This condition corresponds to the maximum or limiting blade loading in the second-stage rotor. This condition of maximum blade loading was investigated in reference 1. The equivalent torque for 100-percent equivalent design speed and equivalent design total to static pressure ratio is approximately 100 percent of the equivalent design value.

SUMMARY OF RESULTS

From an investigation of the over-all performance of a J35-A-23 two-stage turbine, the following results were obtained:

1. Limiting blade loading occurred in the second-stage rotor at equivalent design speed and pressure ratio, which limited the equivalent work output to approximately 95 percent of the design value.

2. The turbine brake internal efficiency obtained at equivalent design speed and pressure ratio was approximately 0.75.

3. A peak turbine brake internal efficiency between 0.81 and 0.82 was obtained at equivalent design speed and a pressure ratio of approximately 2.60.

4. The equivalent weight flow at equivalent design speed and design pressure ratio was approximately 106 percent of the design value.

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APPENDIX - APPROXIMATE METHOD OF DETERMINING EQUIVALENT DESIGN CONDITIONS

Information on the effect of heat capacity lag in turbine nozzles presented in reference 2 indicates that for some typical turbines used in turbojet engines the vibrational energy of the gas molecules is unavailable in the expansion process. This condition corresponds to a constant value of γ of 1.4. For this particular engine, however, examination of the results of reference 1 indicates that the actual flow processes may be more closely approximated by using the equilibrium value of γ because of the high pressure level (compressor pressure ratio of 8.75) and increased length of the flow path through the turbine. Consequently, an average equilibrium value of $\gamma = 1.315$ was selected as a representative value. This value of γ was applied in the following methods of determining the equivalent design points shown on the performance curves.

<u>Determination of equivalent weight flow</u>. - By writing the equation of continuity in terms of critical velocity V_{cr}, area A_{cr}, and density ρ_{cr} and solving for the critical area, the following equation is obtained:

$$A_{cr} = \frac{w_{cr}V_{cr}}{p'g} \frac{1}{\gamma} \left(\frac{\gamma+1}{2}\right)^{\frac{\gamma}{\gamma-1}} \tag{1}$$

The critical area for turbine operating conditions is equated to the critical area for NACA standard sea-level conditions at the turbine inlet to obtain

$$\frac{w_{cr,e}V_{cr,e}}{p'_e} \frac{1}{\gamma_e} \left(\frac{\gamma_e+1}{2}\right)^{\frac{\gamma_e}{\gamma_e-1}} = \frac{w_{cr,0}V_{cr,0}}{p'_0} \frac{1}{\gamma_0} \left(\frac{\gamma_0+1}{2}\right)^{\frac{\gamma_0}{\gamma_0-1}}$$

Solving for the critical weight flow at NACA standard sea-level conditions, the following equation may be written:

$$w_{cr,0} = \frac{w_{cr,e}\dfrac{V_{cr,e}}{V_{cr,0}}}{\dfrac{p'_0}{p'_e}} \frac{\gamma_0}{\gamma_e} \left[\frac{\left(\dfrac{\gamma_e+1}{2}\right)^{\frac{\gamma_e}{\gamma_e-1}}}{\left(\dfrac{\gamma_0+1}{2}\right)^{\frac{\gamma_0}{\gamma_0-1}}}\right] \tag{2}$$

Equation (2) may be written as

$$w_{cr,0} = \frac{w_{cr,e} \sqrt{\theta_{cr}}}{\delta} \beta \tag{3}$$

Figure 6 shows the variation of β as a function of γ.

2200 Determination of equivalent work. - The work output of a turbine rotor-blade row may be expressed by the following equation:

$$E = \eta_i \frac{\gamma}{\gamma-1} \frac{R}{J} T'_1 \left[1 - \left(\frac{p'_{x,2}}{p'_1}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{4}$$

Dividing equation (4) by V_{cr}^2 and simplifying gives

$$\frac{E}{V_{cr}^2} = \eta_i \frac{\gamma+1}{2(\gamma-1)g} \left[1 - \left(\frac{p'_{x,2}}{p'_1}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{5}$$

Equating $E_0/V_{cr,0}^2$ for standard sea-level conditions to $E_e/V_{cr,e}^2$ for engine operating conditions gives

$$\frac{E_0}{V_{cr,0}^2} = \frac{E_e}{V_{cr,e}^2} \tag{6}$$

Combining equation (6) with equation (5) gives

$$\eta_{i,0} \left(\frac{\gamma_0+1}{\gamma_0-1}\right) \frac{1}{2g} \left[1 - \left(\frac{p'_{x,2}}{p'_1}\right)_0^{\frac{\gamma_0-1}{\gamma_0}}\right] = \eta_{i,e} \left(\frac{\gamma_e+1}{\gamma_e-1}\right) \frac{1}{2g} \left[1 - \left(\frac{p'_{x,2}}{p'_1}\right)_e^{\frac{\gamma_e-1}{\gamma_e}}\right] \tag{7}$$

Assuming that the turbine efficiency η_i does not change

$$\left(\frac{\gamma_e+1}{\gamma_e-1}\right)\left[1 - \left(\frac{p'_{x,2}}{p'_1}\right)_e^{\frac{\gamma_e-1}{\gamma_e}}\right] = \left(\frac{\gamma_0+1}{\gamma_0-1}\right)\left[1 - \left(\frac{p'_{x,2}}{p'_1}\right)_0^{\frac{\gamma_0-1}{\gamma_0}}\right] \tag{8}$$

To preserve the equality of equation (8) as γ changes, the pressure ratio must vary. Figure 7 shows the variation of the ratio of the pressure ratio at standard conditions to the pressure ratio at engine operating conditions as a function of γ for various constant values of each pressure ratio.

REFERENCES

1. Hauser, Cavour H., Plohr, Henry W., and Sonder, Gerhard: Study of Flow Condition and Deflection Angle at Exit of Two-Dimensional Cascade of Turbine Rotor Blades at Critical and Supercritical Pressure Ratios. NACA RM E9K25, 1950.

2. Spooner, Robert B.: Effect of Heat-Capacity Lag on a Variety of Turbine-Nozzle Flow Processes. NACA TN 2193, 1950.

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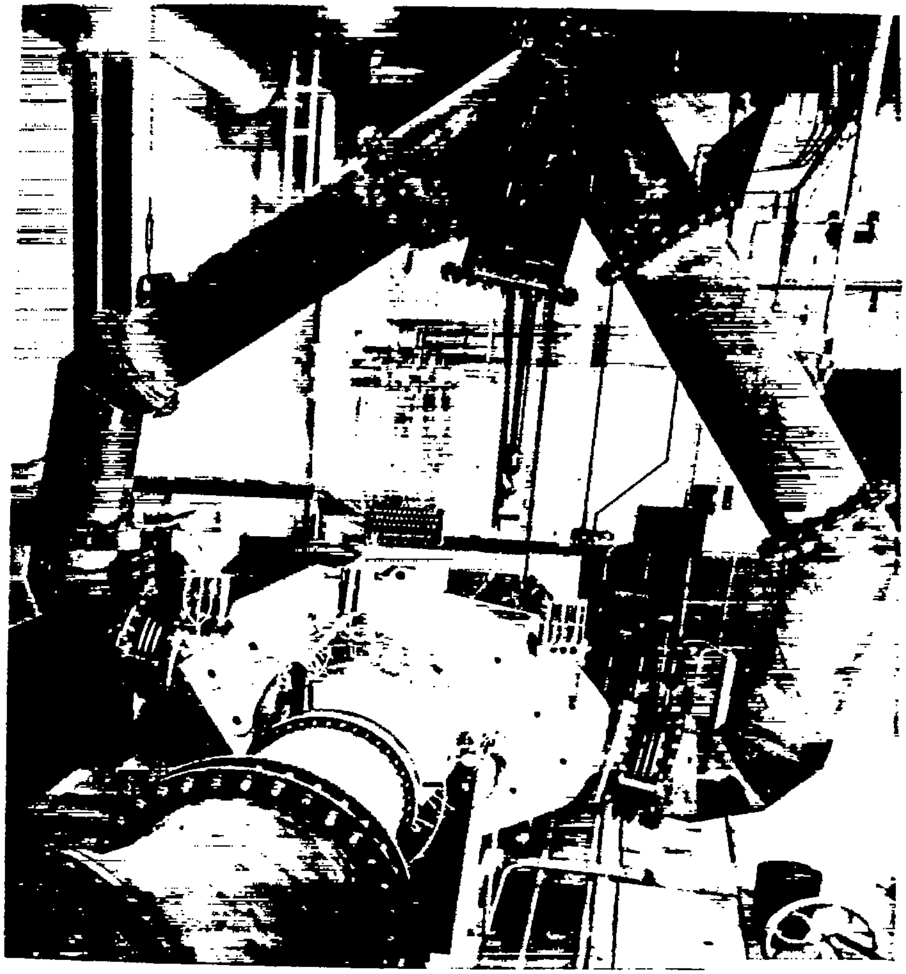

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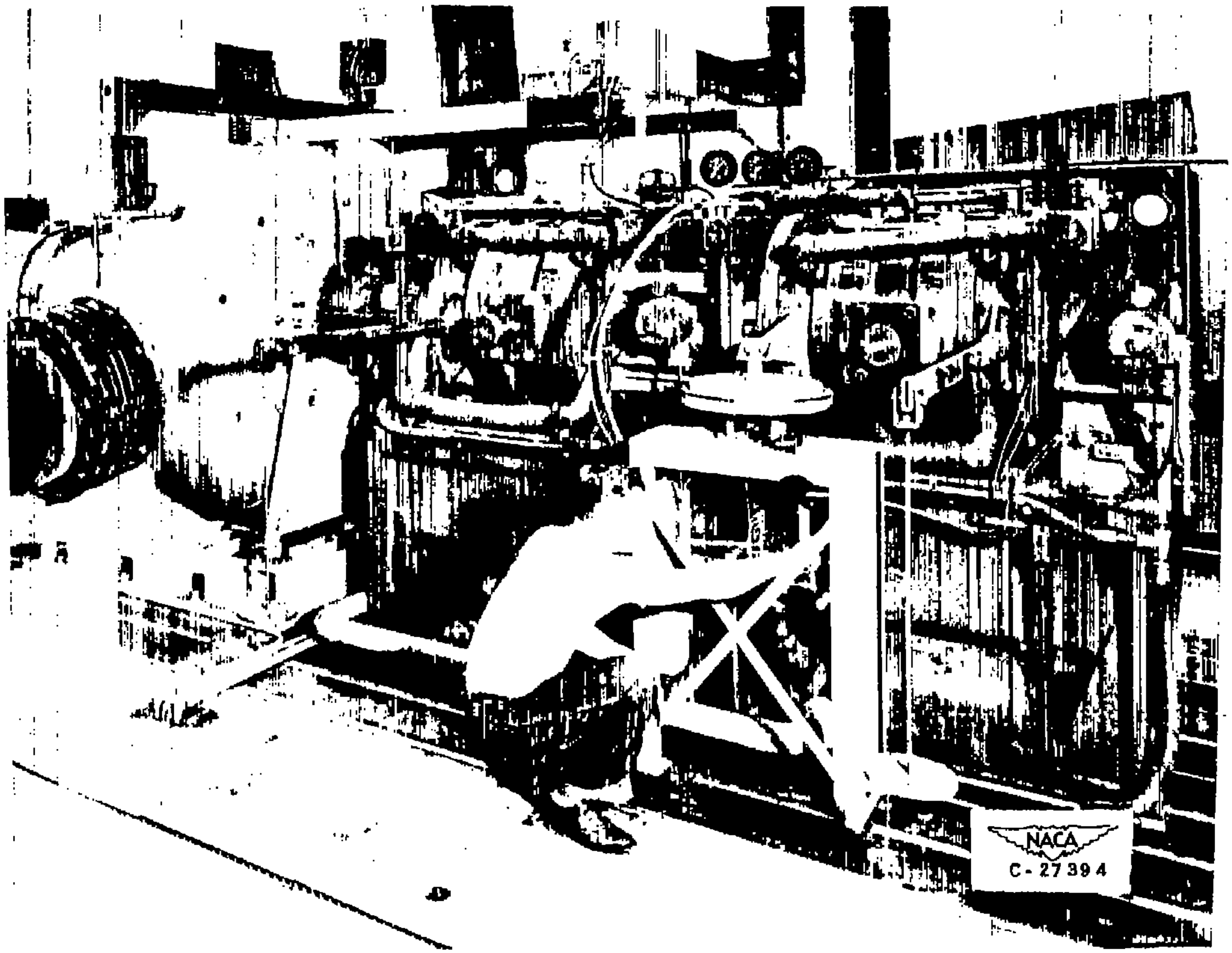
(a) Rear view showing inlet pipes to plenum chamber, turbine, and exhaust pipe.

Figure 1. - Installation for experimental investigation of J35-A-23 two-stage turbine.



(b) Front view showing dynamometers and thrust meter.

Figure 1. - Concluded. Installation for experimental investigation of J35-A-23 two-stage turbine.

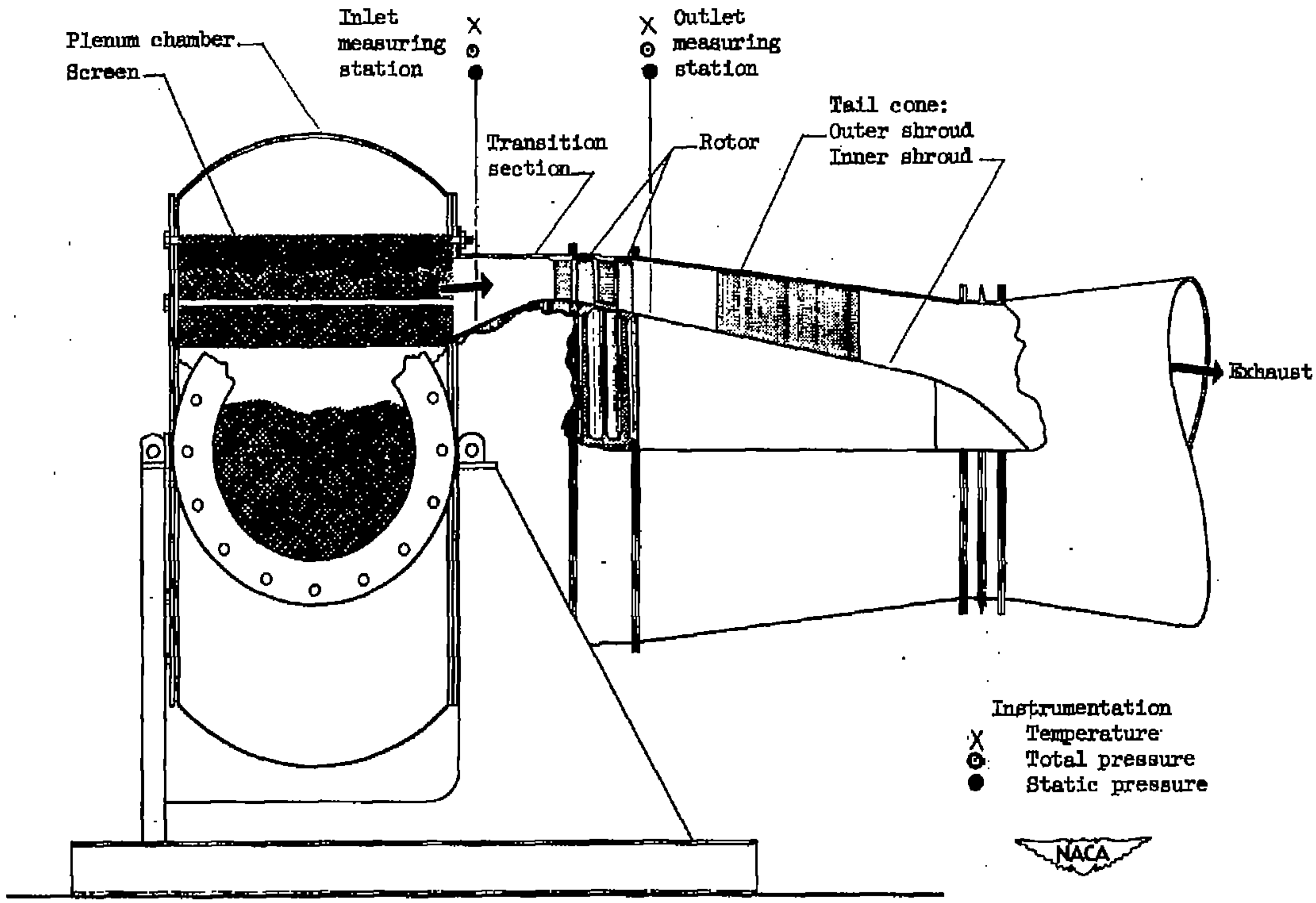


Figure 2. - Schematic diagram of turbine assembly and instrumentation.

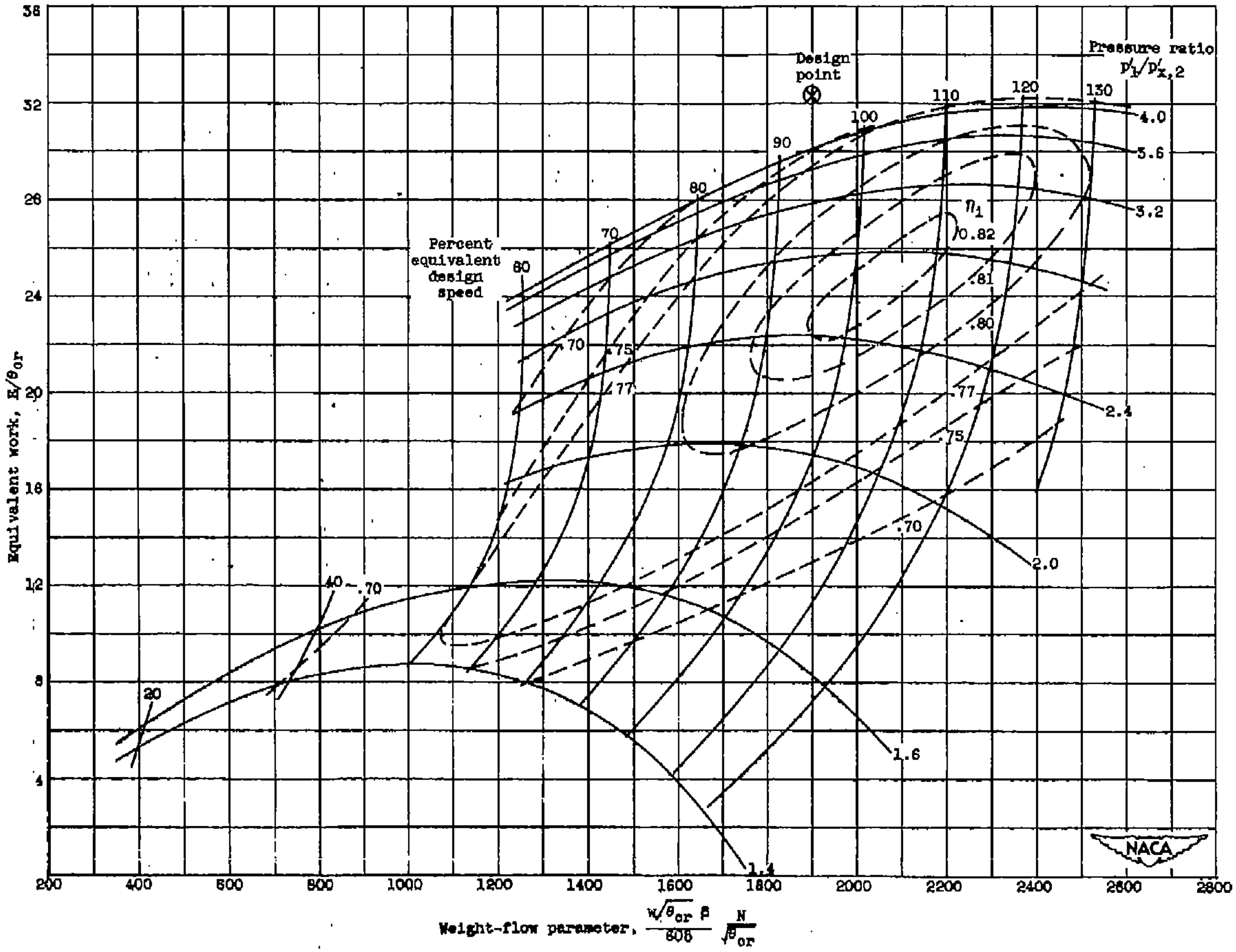


Figure 3. - Over-all performance of J35-A-23 two-stage turbine presented in terms of equivalent work and a weight-flow parameter for lines of constant equivalent speed, pressure ratio, and brake internal efficiency. Equivalent design speed 3035 rpm.

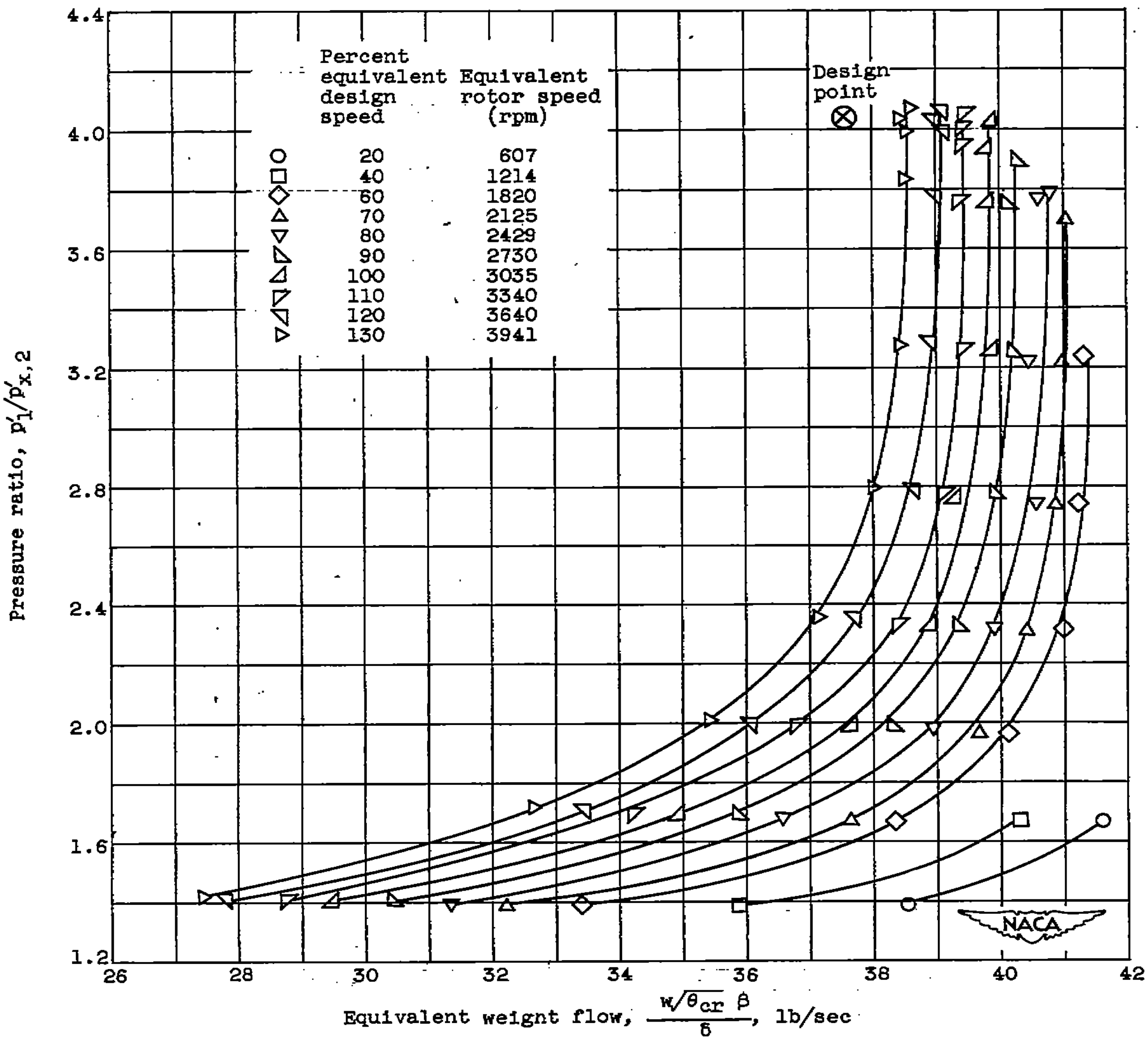


Figure 4. - Variation of equivalent weight flow with pressure ratio for various turbine speeds.

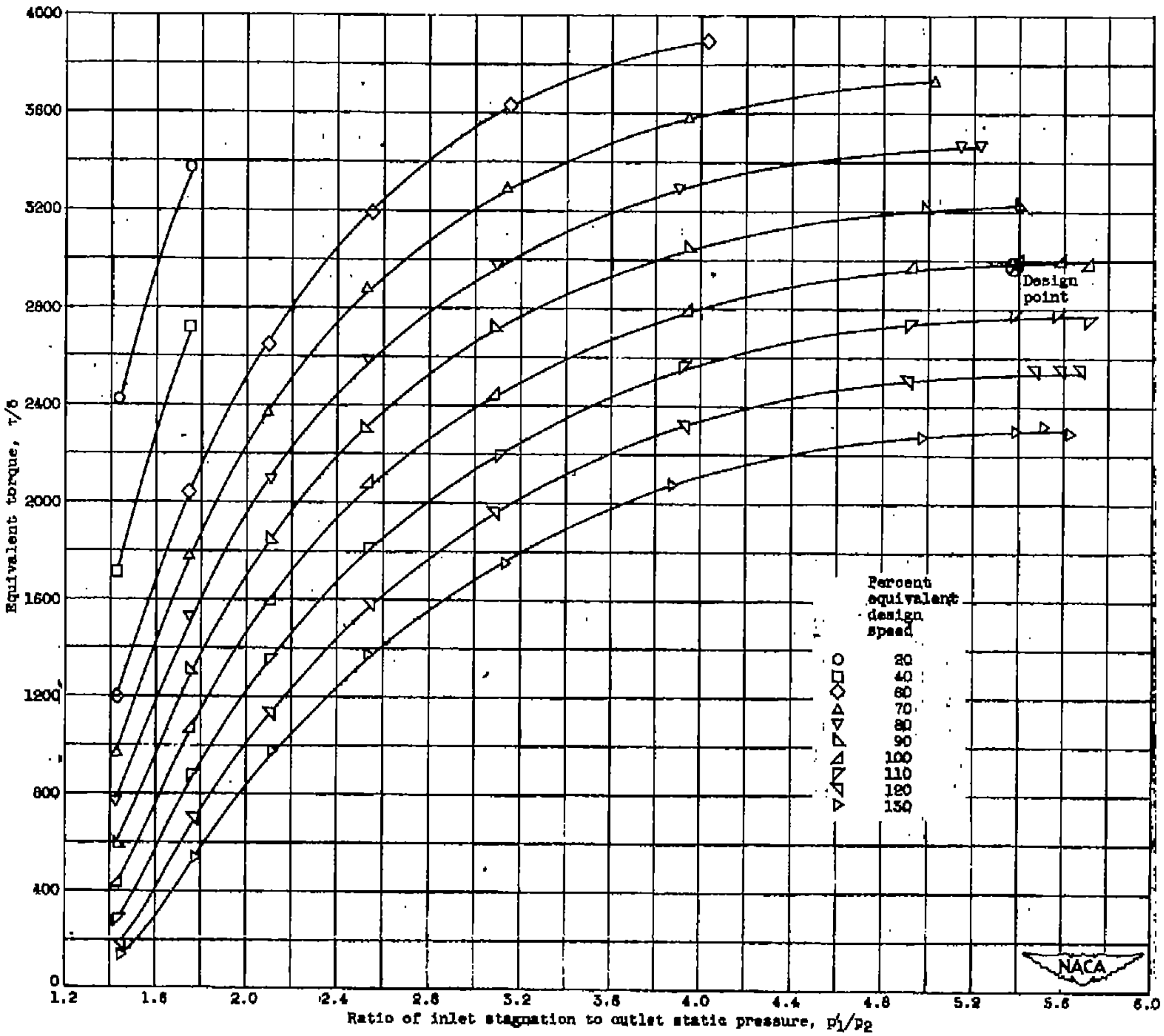


Figure 5. - Variation of equivalent torque with turbine pressure ratio for various turbine speeds.

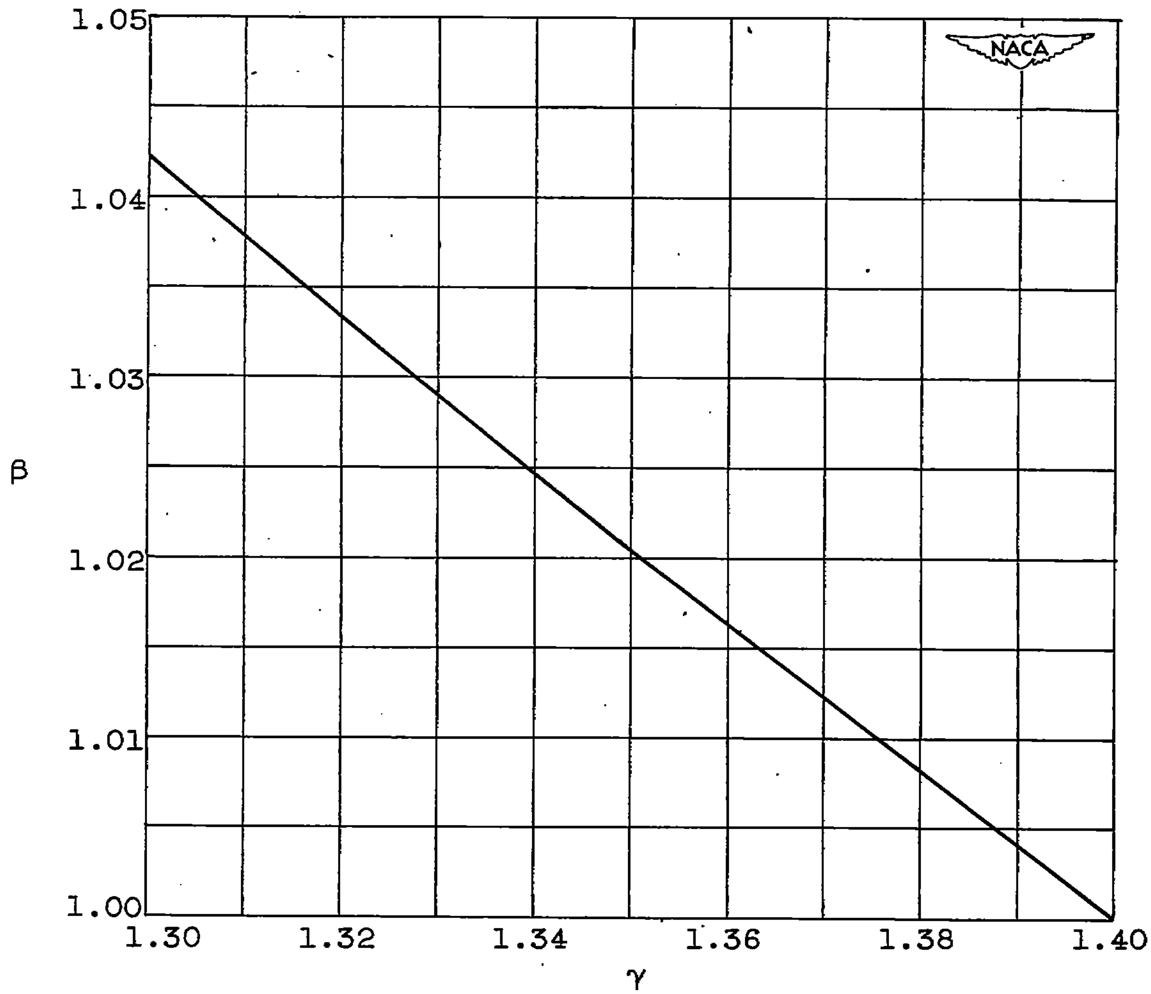


Figure 6. - Variation of β as a function of γ.

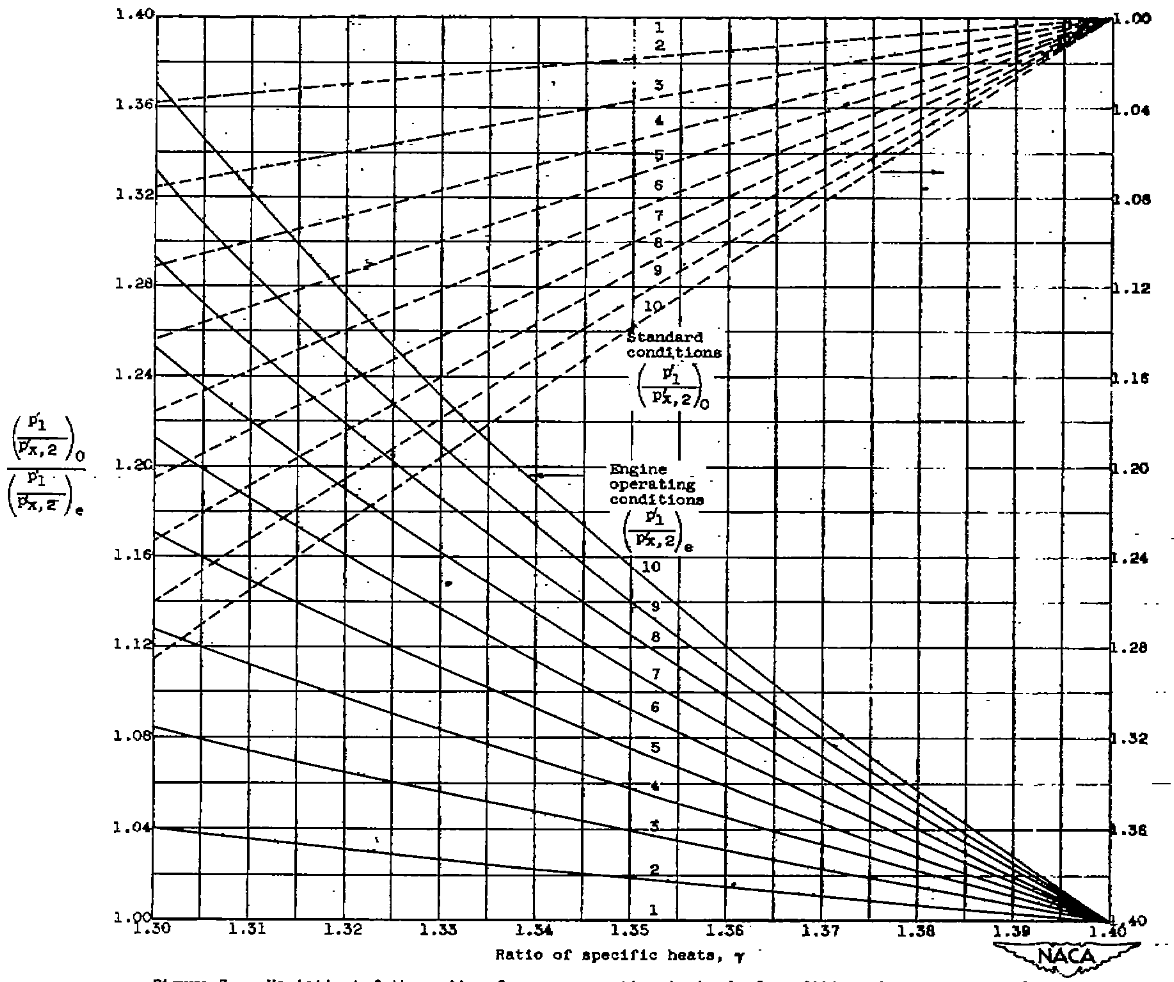


Figure 7. - Variation of the ratio of pressure ratio at standard conditions to pressure ratio at engine conditions with ratio of specific heats.

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SECURITY INFORMATION

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